DEVELOPMENT OF LOW-NOISE HIGH-PRESSURE FANS FOR SHIP COLLECTIVE PROTECTION II: ENGINEERING DEVELOPMENT

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ABSTRACT

The focus of development of these high-pressure vaneaxial fans for ship collective protection systems (CPS) was on redesigned impellers and stators to increase aerodynamic efficiency and lower the acoustic noise levels produced by the fans. The sound power generated by these fans was been reduced by over 12 dB and the energy consumption by over 20%. The rotors have been designed and fabricated to comply with new factor of safety requirements being proposed for all vaneaxial CPS fans. To validate the suitability of the design for ship applications, a high-impact shock test was conducted with the fan operating at 3,600 rpm. Strain measurements from fan rotor blades were correlated with FEA predictions to validate the design.

1. INTRODUCTION

Collective protection systems (CPS) in US Navy ships use high-pressure vaneaxial fans to draw outside air through CBR filters and pressurize interior compartments. The fans are capable of creating pressures of > 14 inches of water, gauge in order to overcome the pressure drop through the filter banks and supply the required 1 to 2 inch water, gauge overpressure in protected zones. A longstanding complaint of CPS on Navy ships has been the noise generated by these fans, especially when they are located near berthing or operational spaces. This paper focuses on engineering development of a new generation of A104 (3600 cfm) and A105 (5400 cfm) vaneaxial fans with greatly reduced noise and energy consumption. The new fans are currently being installed in amphibious ships (LHD-1, LHA-1, and LSD-44 Classes). The fans are based on a design developed and patented[1,2] by the Naval Surface Warfare Center (NSWC) Carderock and Dahlgren Divisions. The fans are currently being manufactured by American Fan Company of Fairfield, Ohio

The basic layout of the new Annapolis-design fans is shown in Figure 1. The rotor for the R&D version of the fans uses a separate nose cone, and this design was also used for the production version. The A104 and A105 rotors have a 21.094 inch outer diameter. The clearance between the tip of the rotor blades and the rotor race is 0.014 inch to 0.020 inch. The rotors have 13 unequally spaced blades that

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have varying overlap between blades (see Figure 2). Note that Figure 1 shows a fan housing with a separate rotor race, which is a design used by American Fan, but was not used in the original R&D fans.

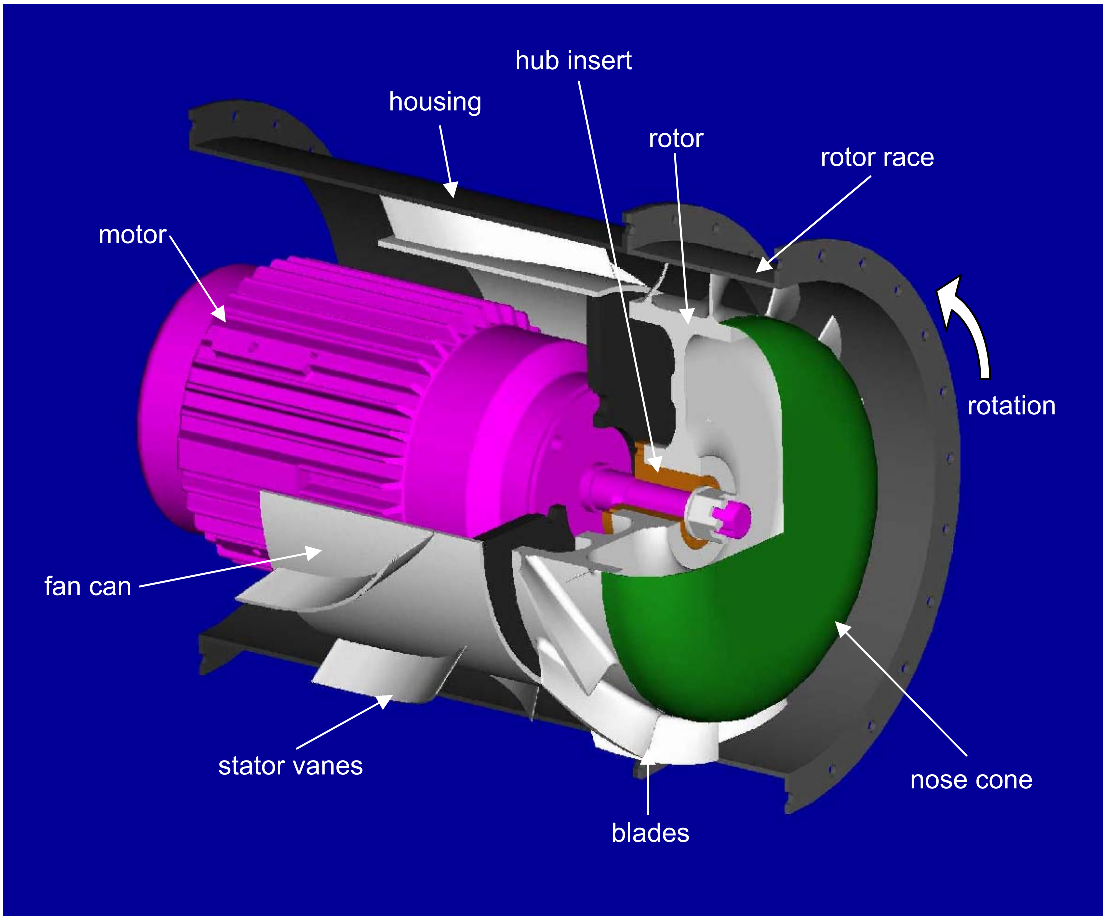

Figure 1. Basic layout of low-noise CPS fans.

Section 2 of this paper discusses the redesigned impellers and stators used to increase aerodynamic efficiency and lower noise.[3] In Section 3 performance measurements are shown that compare noise and air flow performance for the new designs and the previous designs. In addition to the performance improvements of the new fans, the rotors have been designed and fabricated to comply with new factor of safety requirements being proposed for all vaneaxial CPS fans.[4] Results of extensive finite element analyses (FEA) performed on the impellers to support the designs are shown in Section 4 and recommendations for the factors of safety to be amended to the performance specification governing CPS fans[5] are reported in Section 5. In Section 6, conclusions and recommendations based on the work performed are described. Please note that, since this paper describes a multi-year effort that has been described extensively in other reports, significant content has been excerpted from those reports.[3,4,6,8,9] While every effort has been made to properly cite those sources at some point in this paper, they are not cited in every case in order to avoid an excess of clutter.

2. IMPROVED AERODYNAMIC DESIGN

The basic layout of the Annapolis fans is shown in Figure 1. The rotor for the R&D version of the fans used a separate nose cone, and this design was also used for the production version. The A104 and A105 rotors have a 21.094 inch outer diameter. The clearance between the tip of the rotor blades and the rotor race is 0.014 inch to 0.020 inch. The rotors have 13 unequally spaced blades that have varying overlap between blades (see Figure 2 and Figure 3). Note that Figure 1 shows a fan housing with a separate rotor race, which is a design used by American Fan, but was not used in the original R&D fans.

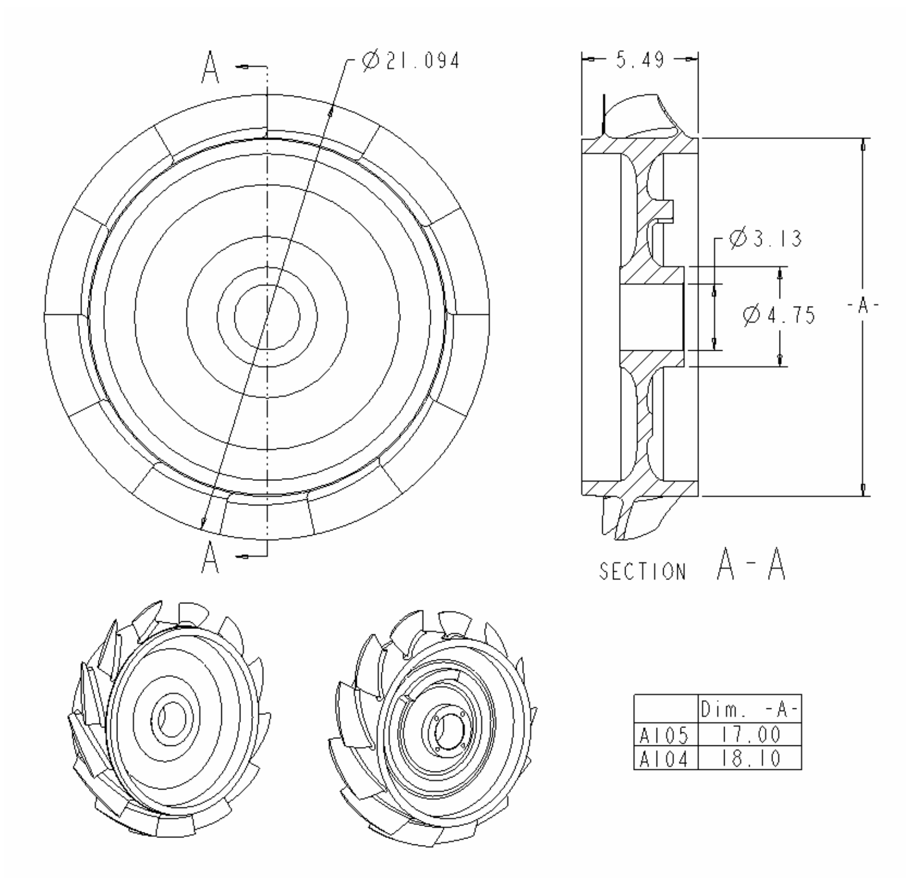

Figure 2. Geometry of Annapolis-design rotors.

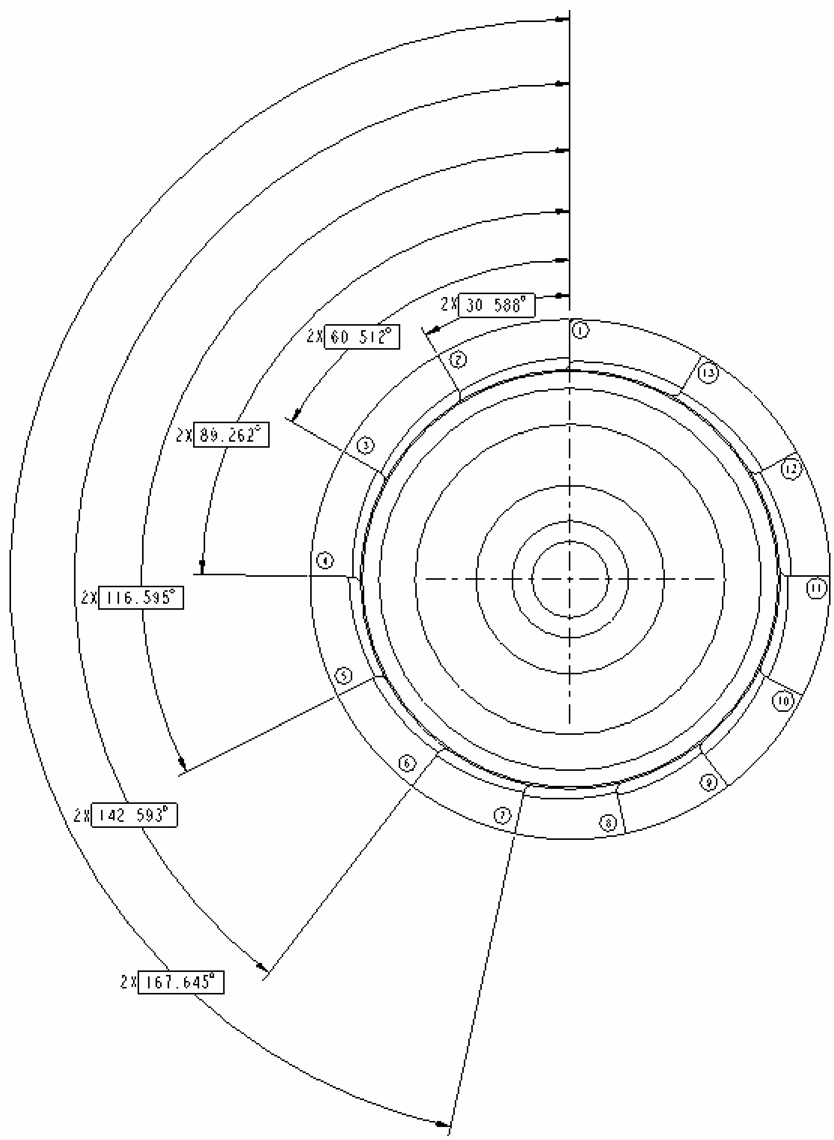

Figure 3. Angular placement of rotor blades.

The rotor blades use a NACA-65 airfoil design with a C-V nose. The blades have a specific lean and "twist" radially outward from the hub. A "root radius" at the base of the blades where they intersect the hub reduces stresses. As is shown in Figure 2, when viewed from the front of the rotor the unequally-spaced blades overlap. This is a performance enhancement that made fabrication more difficult, especially for cast rotors where a two-piece mold cannot be used.

The design of the stator vanes also used an airfoil nose cone, but the body of the vane can be formed out of flat plate. Both the twist and the airfoil nose on the stator vanes are features that add cost because of either additional tooling, processing time, or both.

A view of the geometry for the stator vanes and rotor blades is given in Figure 4. The blades and vanes have a specific twist that was chosen for improved aerodynamic performance and lower noise (see

Figure 5). The geometry was chosen to minimize aerodynamic separation and turbulence, thus increasing the efficiency and reducing generated noise. The relatively complex geometry for the blades and vanes was perhaps the most challenging part of manufacturing the fans.

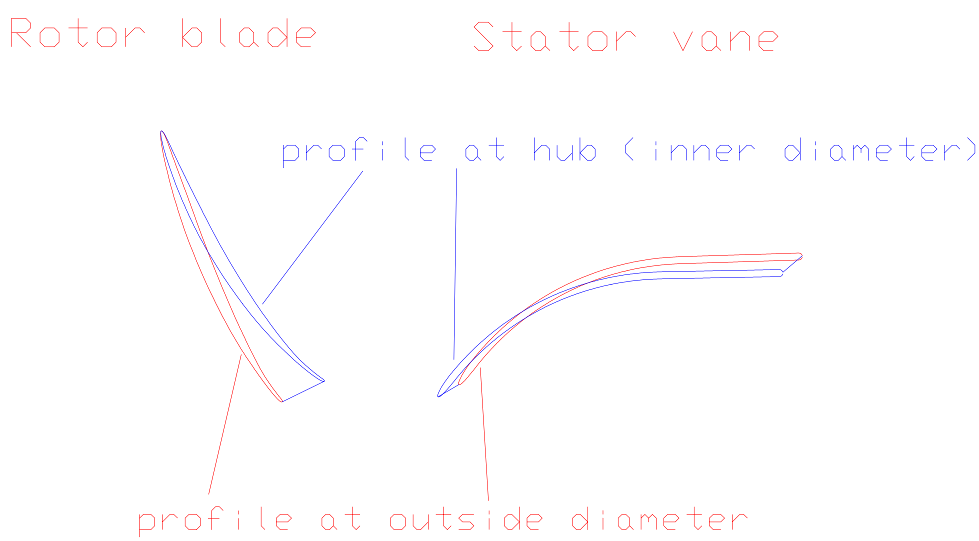

Figure 4. Cross-sections of rotor and stator blades at inner/hub (blue) and outer/tip (red) diameters. Leading edges are at left sides of both blades.

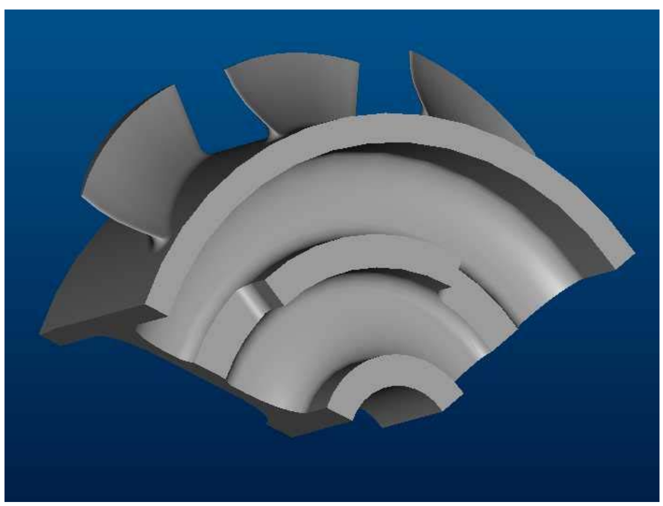

Figure 5. Blade trailing edge view.

3. PERFORMANCE MEASUREMENTS

Table 1 shows performance of fans based on the Annapolis design compared to the R&D version and previous-generation A104 and A105 CPS fans. Note the dramatic reduction in sound power levels. If the geometry of the sound field for the Annapolis and previous design fans were similar, this sound power reduction could give a sound pressure reduction of roughly 26 dB in spaces where the fans were installed. The human perception of this noise reduction would be that the Annapolis-design fan is ¼ to 1/8 the loudness of the previous design. The measured peak power is reduced 23% for the A104 and 20% for the A105 compared to previous CPS fans. The reduction in peak power is also significant, as it allows a reduction in motor size for the A105 from 20 hp to 15 hp. Since a 12 hp motor is not available, the motor size for the A104 does not change. The energy reductions will also reduce life-cycle costs associated with electric power significantly.

Table 1. Comparison of production fan performance to R&D fan performance.

<table>
<tr><th colspan="2" rowspan="2"></th><th>Peak Horsepower</th><th>Motor Size</th><th>Avg. Sound Power</th></tr>
<tr><th>(Hp)</th><th>(Hp)</th><th>(dB re 10^{-12} W)</th></tr>
<tr><td rowspan="3">A104</td><td>Previous Design</td><td>15.5</td><td>15</td><td>116.3</td></tr>
<tr><td>Annapolis Design</td><td>11.6</td><td>12</td><td>103.2</td></tr>
<tr><td>American Fan implementation of Annapolis Design</td><td>11.9</td><td>15</td><td>102.5</td></tr>
<tr><td rowspan="3">A105</td><td>Previous Design</td><td>19.2</td><td>20</td><td>118</td></tr>
<tr><td>Annapolis Design</td><td>15.6</td><td>15</td><td>105</td></tr>
<tr><td>American Fan implementation of Annapolis Design</td><td>15.4</td><td>15</td><td>104.5</td></tr>
</table>

Figure 6 and Figure 7 show the results of sound power measurements performed at Woods Air Movement Limited of Colchester, United Kingdom. Note that these tests deviate somewhat from the AMCA 300 test required by Mil-F-24755(SH). The 63 Hz band is the only octave band in which the deviation should be significant. The Woods measurements should give a sound power that is roughly 5 dB higher in the 63 Hz band than would be measured using AMCA 300 procedures.[7]

As can be seen in the figures, the sound power levels are significantly lower than those required by Mil-F-24755(SH). Thus, the goal of creating low-noise fans has been met.

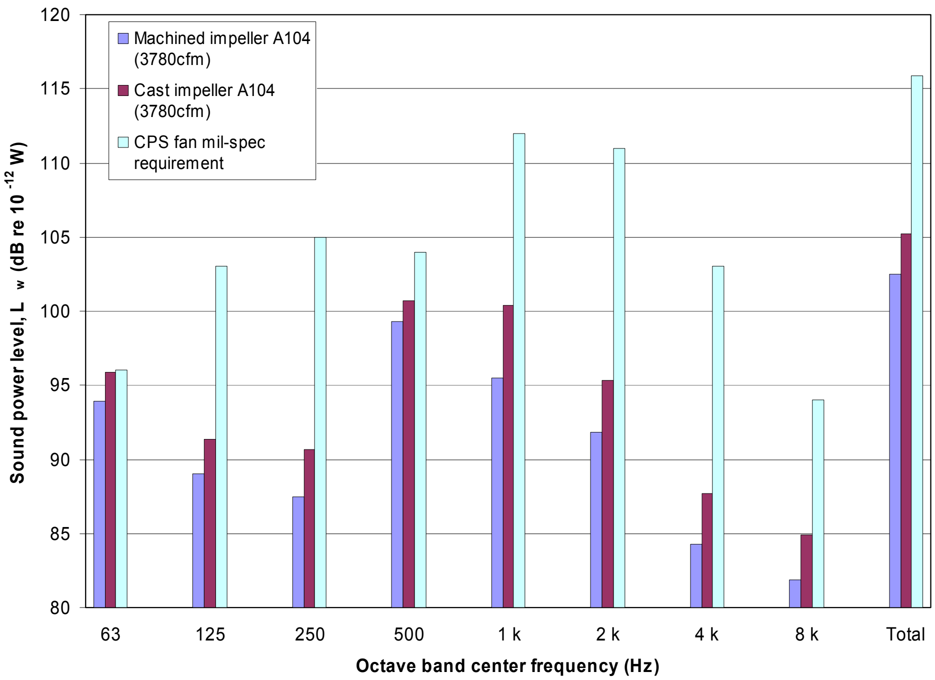

Figure 6. Measured sound power levels for A104 fan with cast and machined rotors.

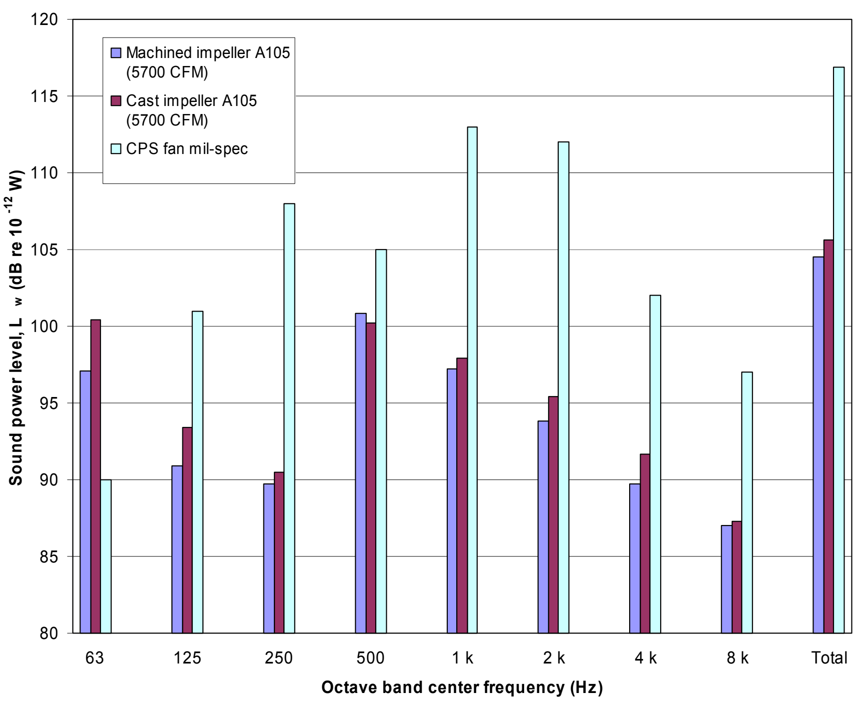

Figure 7. Measured sound power levels for A105 fan with cast and machined rotors. Note that test facility was not set up to get accurate measurements of the sound power in the 63 Hz band.

Each production fan was tested to AMCA 210 specifications (Figure 12) at American Fan's facility in Fairfield, Ohio (see Figure 8). Note that only the flow performance was tested at American Fan. Noise levels were obtained from the tests run on fans of the same configuration at Woods Air Movement in Colchester, England. The noise tests run at Woods used two of the machined rotors (one A104 and one A105) that were later installed in the first production fans, but had different housings.

Figure 8. A104 fan installed in test duct with bell-mouth on inlet.

Figure 9 shows the flow performance of the first production Annapolis-design A105 fan. The fan characteristic curve passes squarely through the performance requirements box. Previous tests at Woods had determined that the peak power of 15.4 hp (for a Reliance motor rated at 15 hp) does not adversely affect motor heating.

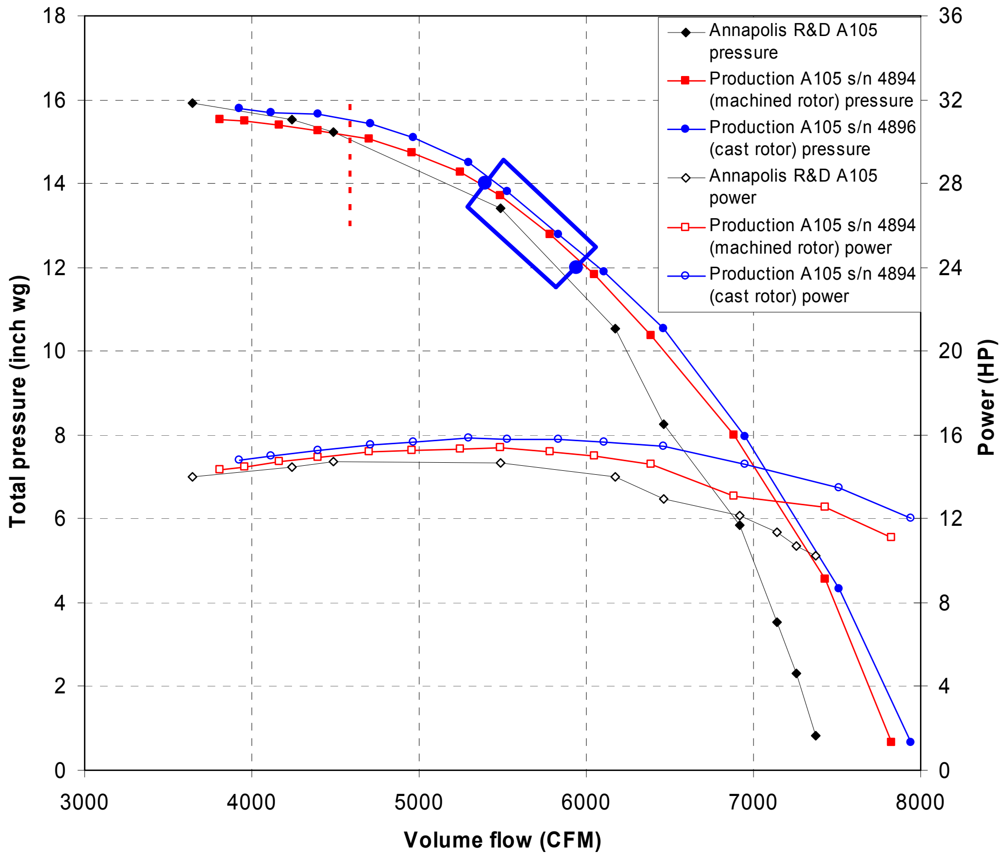

Figure 9. Performance of first production A105 fans and original R&D fan. Compared to the R&D version, the production fans have (for both machined rotor and cast rotor versions) a smaller hub (17.00" vs. 17.25") and larger blade root radius (0.325" vs. 0.188").

As can be seen in Figure 10, the flow performance of two of three of the first production Annapolis-design A104 fans were outside the specified performance window for flow rates above about 3900 CFM. This performance issue was apparently later corrected with modifications to the manufacturing processes.

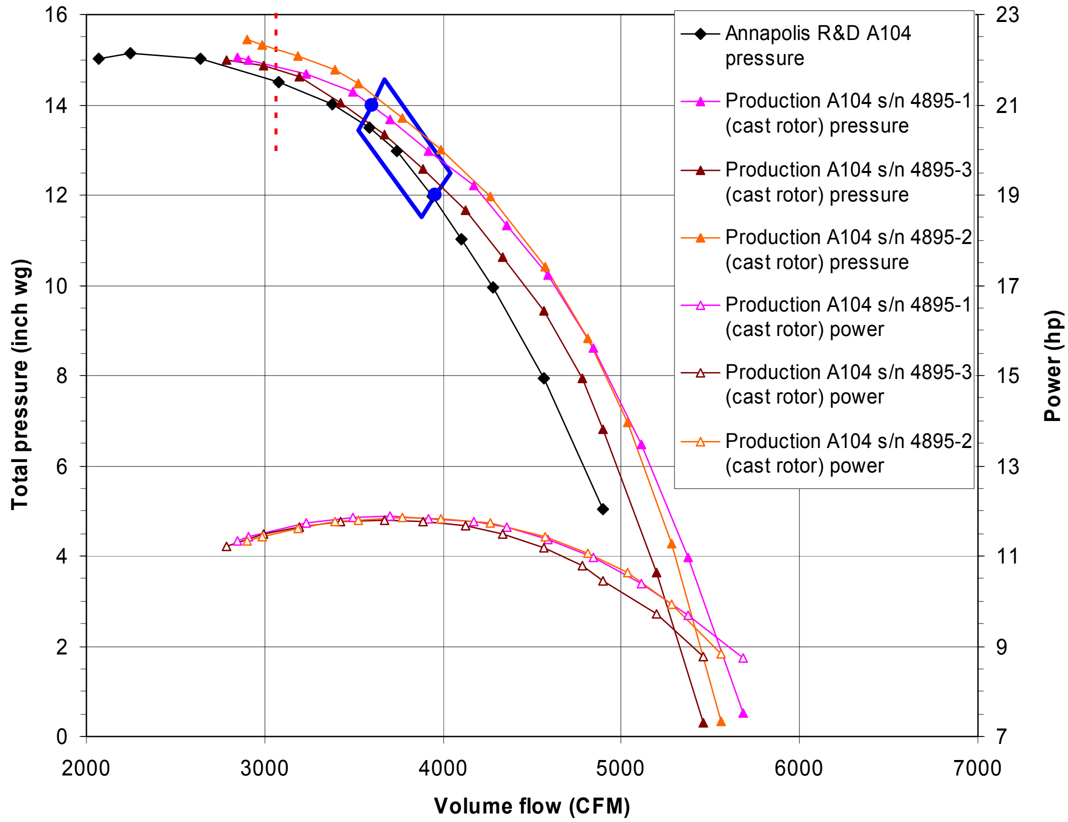

Figure 10. Performance of first cast-rotor production A104 fans.

4. ROTOR MECHANICAL EVALUATION

In this section we describe analytical and experimental evaluations of the stresses in the fan rotors. In Section 4.1 the stress analyses performed to ensure that the rotors for the Annapolis A104 and A105 fans meet the requirement of a factor of safety of 8 that was specified at the time the fans were designed. In Section 4.2 results of an instrumented shock test on an A105 prototype are reported.

4.1 Stress Analyses

Design work was performed under the assumption that the rotors will ultimately be cast from aluminum alloy rather than machined from billet. Fan rotors machined out of 6061-T6 aluminum to the geometry specified by this report will have factors of safety of about 11.

Stress analyses performed in this report used both static (i.e., non-time-varying) and dynamic (i.e. shock) loads. For static tests the primary loading was from the rotational velocity of the fan rotors. Pressure loading of the fan blades was also studied, but this load had a minor impact on the peak stresses (which were typically about 3% lower when the pressure load was imposed in addition to the centrifugal load). Results of analyses for the dynamic case are reported elsewhere.[4]

FEA runs were made using both full 13 blade models, and models taking advantage of cyclic symmetry.[8] The symmetric models only had 6 or 12 blades (one per 60 degree symmetric segment, or 3 per 90 degree segment). The stress variations caused by the changes in blade spacing were only about 2%. The advantage of using the cyclic symmetry models is that they require only 1.5 hours to run, versus 6 hours for the full models.

Based on the FEA work, the final geometry for the production A104 and A105 rotors was created. The basic A105 rotor is 21.094 inch in diameter, has a hub diameter of 17.00 inch (compared to the R&D version, which used 17.25 inch), and has 13 unequally spaced blades (see Figure 3). The final weight of the A105 rotor is 41.3 lb. In order to meet the factor of safety of 8 for the cast A105 rotor, the following design characteristics were used:

- Blade root radius = 0.325 inch (R&D version used 0.188 inch)
- Hub thickness = 0.75 inch (the R&D version used 0.50 inch)
- Rotor balance is achieved with a balancing ring as shown in Figure 2 (the R&D version did not include this feature).
- No holes are allowed anywhere on the rotor for balancing purposes.

The A104 rotor is 21.094 inch in diameter, has a hub diameter of 18.10 inch (compared to the R&D version, which used 18.22 inch), and also has 13 unequally spaced blades (see Figure 2). The final weight is 36.4 lb. Because the blades are shorter than on the A105 rotor, the stresses tended to be lower, so no variations from the original blade geometry were required. In order to meet the factor of safety of 8 for the A104 rotor, the following design characteristics were used:

- Blade root radius = 0.188 inch (same as the R&D version)
- Hub thickness = 0.50 inch (same as the R&D version)
- Rotor balance is achieved with a balancing ring as shown in Figure 2 (the R&D version did not include this feature).
- No holes are allowed anywhere on the rotor for balancing purposes.

As a final check on the balancing ring an analysis of the full rotor was completed, without using cyclic symmetry. The correct blade spacing was used with the added balancing features. For this analysis both the centrifugal and pressure loads were included. As can be seen in Figure 11, peak stresses were within acceptable levels for the required FOS of 8. Also note that there are no unbalanced stresses around the inner bore through the hub (this means that the rotor has indeed been balanced with the addition of the balancing features).

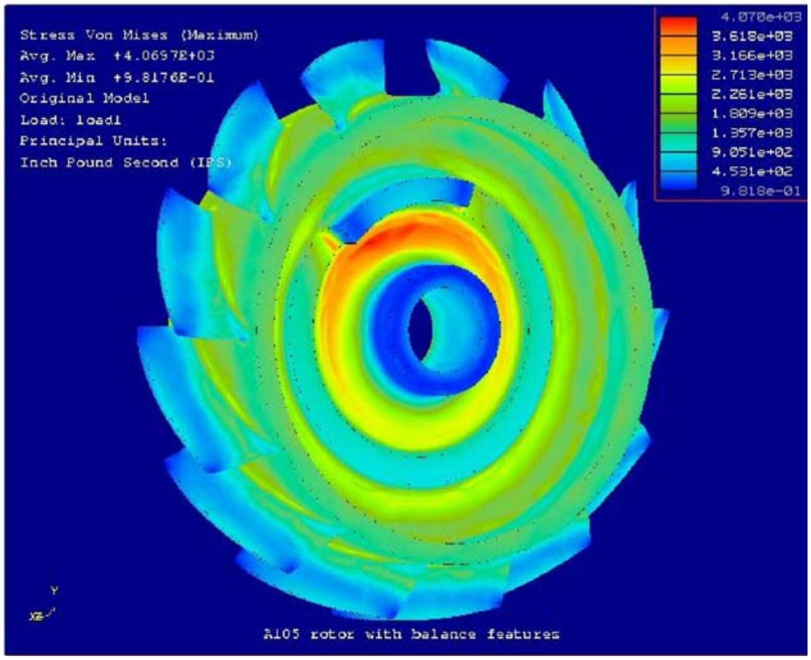

Figure 11. View of discharge side of full A105 rotor with unequally spaced blades and balance features. Loads were both rotation and pressure. Peak stress = 4.1 ksi, FOS=8.

In addition to the FEA described above, stress analyses were performed to study the effect of manufacturing tolerances and defects on the rotors. These effects were then used to develop the factors of safety described in Section 5 of this report.[6,9]

Ana example of an evaluation of a manufacturing defect is given here: For a load of a defined magnitude, the stress induced in structures such as the fan rotor will vary if the geometry of the structure varies. The geometry may vary as a result of manufacturing tolerances and the size of pores that occur in castings.

By knowing the geometric variation associated with the manufacturing process used we can estimate the factor of safety associated with the geometry. Note that, if there are several ways in which the geometry may vary, we must calculate a factor of safety for each.

Figure 12 shows that a 0.120" pore breaking through the surface of the blade gives a peak stress of 13.5 ksi. Note that this peak stress is localized to a very small region around the knife-edge where the pore breaks the surface of the blade. If a pore of this type were present in a casting it would be likely that the edge would not be sharp, but would have some small curvature.

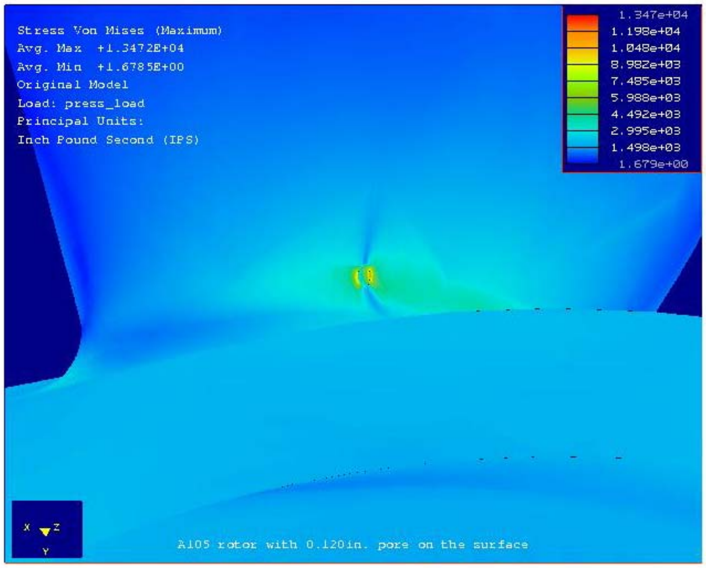

Figure 12. Close-up of blade on A105 rotor with sharp-edged 0.120" pore breaking surface of blade. Peak stress = 13.5 ksi.

Expected stress variations derived from stress analyses such as that in Figure 12 were then used to develop components of a new factor of safety (see Section 5). Table 2 shows the use of the pore analysis from Figure 12 to create the “pore” FOS in the first row. Note that the peak stress in the rotor *without* any pores was about 4.1 ksi, so the stress magnitude of 13.5 ksi increases the total stress by a factor of 3.25, which is then the FOS that must be used if a defect such as that in Figure 12 is present in a cast rotor.

Table 2. FOS n_g for cast rotors

Inspection method	accept/reject criteria => flaw that may be present in finished rotor	FOS, n_g			comment
		tolerance	pore	composite	
Visual	any pore, including slight surface penetration	1.10	3.25	3.58	unacceptable inspection method
Visual + dye penetrant	pore with no surface penetration, assume pore size < 0.120"	1.10	2.20	2.42	minimum inspection
Visual + dye penetrant + x-ray	subsurface pore, 0.120" or less in diameter, not close to blade surface	1.10	1.50	1.65	good inspection method
Visual + dye penetrant + x-ray	subsurface pore, 0.060" or less in diameter, any location	1.10	1.25	1.38	better inspection method
Visual + dye penetrant + x-ray	subsurface pore, 0.060" or less in diameter, reject if pore in high-stress region	1.10	1.10	1.21	depends on expertise of person viewing x-ray; method may be unreliable

4.2 Instrumented Shock Test

To validate the suitability of the fan and rotor designs for ship applications, a high-impact shock test was conducted with the fan operating at 3,600 rpm. Custom test fixtures were designed to obtain shock and strain data from the impeller during normal rotation and during shock events. Strain measurements from fan rotor blades were correlated with FEA predictions to validate the design.[4]

Performance of the rotor under shock is especially important because the rotor blades impact the fan housing. Since the stresses in the rotor under shock are much higher than for any other condition it was critical to evaluate shock when developing the new FOS recommendations. Here we describe both the experimental evaluation of the strain in the rotor under impact and finite element analyses of the stresses that should be present based on the experimental strains.

The shock tests were run at NU Laboratories in Clinton, New Jersey on August 21 and 22, 2001. The A105 fan used was built by American Fan. All shock testing was run with the fan in the horizontal position, as it was believed that this would result in the peak stresses in the rotor. The basic shock procedure used was that given in MIL-PRF-24755A and MIL-S-901D, although additional intermediate hammer drop heights were added. Photographs of the experimental setup are shown in Figure 13 through Figure 17.

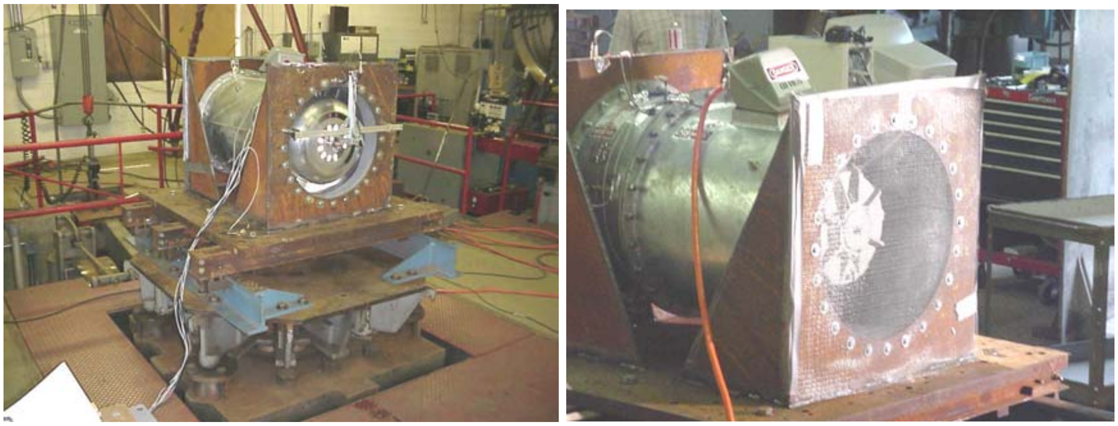

Figure 13. Annapolis-design fan mounted on Navy medium-weight shock test machine.

Figure 14. Mounting of shock accelerometer and strain gauges.

Figure 15. Rotor instrumentation.

Figure 16. T-rail and 10-conductor slip ring.

Figure 17. Mounting of motor accelerometer (left) and strain gauge and accelerometers on rotor race (right).

Figure 18 shows an overlay of two strain gauge signals plus the angular position of the rotor during test 7. The impacts of the two blades (which were approximately 120 degrees apart) occur at different times, but each blade impacts roughly once per revolution of the rotor.

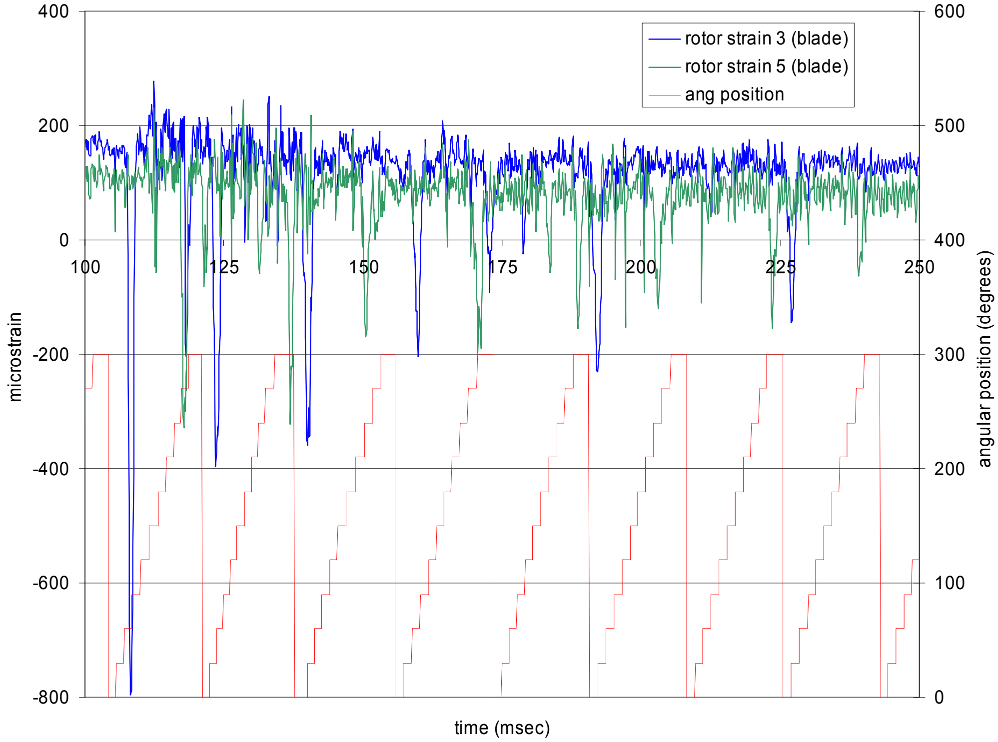

Figure 18. Rotor blade strain and angular position of rotor during shock. Fan was operating during test.

Figure 19 shows a comparison of the signal from a strain gauge on a rotor blade. Because the fan was not running for either of these tests the rotor could be carefully positioned so that an instrumented blade was at bottom-dead-center and would take the initial impact. The hammer drop height was 1 ft for the first test and 2.25 ft the second. The figure shows that both tests resulted in multiple impacts of the rotor with the race, but the higher hammer height did *not* result in a higher strain magnitude. In fact, the lower drop height actually resulted in a higher peak strain.

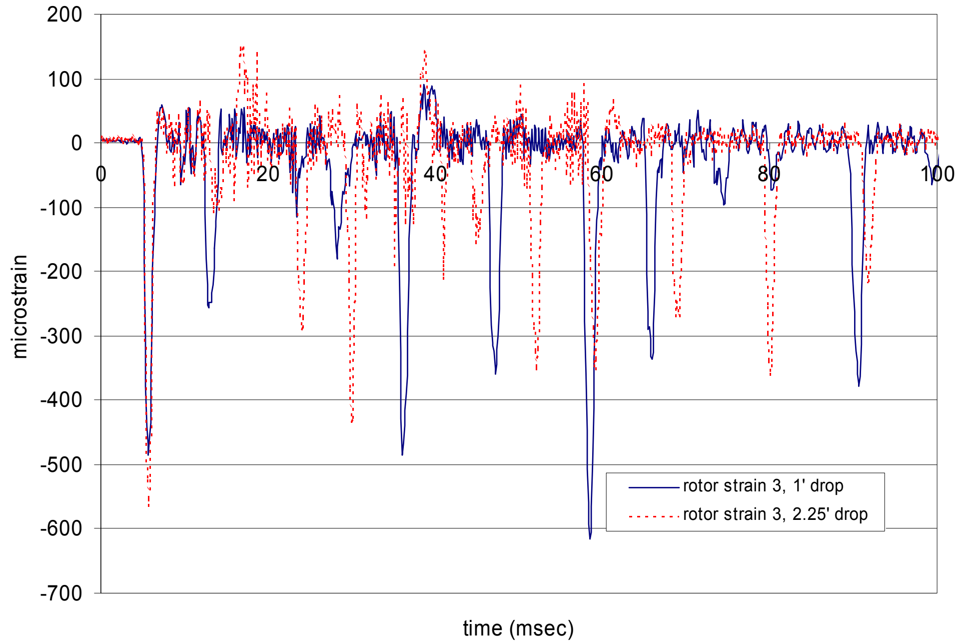

Figure 19. Comparison of strain in rotor blade for 1' and 2.25' hammer drop heights. Fan was off for both tests and strain gauge 3 was at bottom-dead-center.

Using the strain values measured at specific locations on the rotor, FEA was then used to correlate these strains with predicted maximum stresses under shock (see Figure 20). The impact factor of safety described in the next section is based on these measurements and analyses.

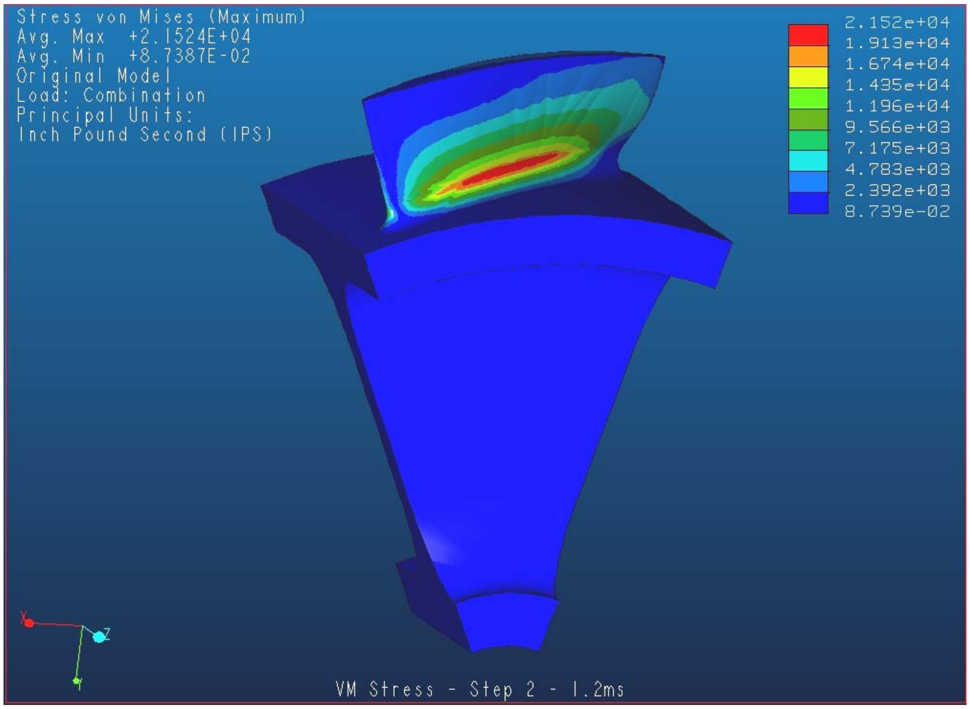

Figure 20. Stress calculated for dynamic impact loads on A105 rotor. View of front of rotor.

5. FACTOR OF SAFETY

The discussion contained in this section grew out of New World's work on the mechanical design of rotors for the low-noise Annapolis designs of the A104 and A105 fans.[8] During the design process, we were forced to make mechanical changes (thicker hub and larger blend radii at the base of the fan blades) to the A105 rotor that both added mass and degraded the stall characteristics of the fans. The reason the design changes were adopted, even at the cost of degraded performance, was to meet a factor of safety of 8 for the fan rotors. We are still left with the question "is a factor of safety of 8 necessary?"

Currently the vaneaxial fans are covered by MIL-PRF-24755A. The required factor of safety quoted in MIL-F-24755A section 3.17 is 8. The specific text from section 3.17 of the specification is:

> *3.17 …The total impeller shall have at least a safety factor of 8, based on the ultimate tensile strength of the material…*

Note that the performance specification does not specify the loading condition used for determination of the factor of safety. Based on discussions with NSWC personnel Michael Slipper and John Larzelere, and discussions with representatives of fan companies, the assumption used for the factor of safety calculation is that the fan is operating at its design speed and aerodynamic performance.

Based on the design analyses for the A104 and A105[8] and the shock test performed on the A105 prototype[4] the following recommendations were developed and included in a proposed amendment to MIL-PRF-24755A[9] to supersede the current FOS requirement of 8 given above.

Table 3. Required factor of safety for normal operating conditions.

Impeller type	Load	$n_{rotation}$
Impeller machined from billet or plate	Rotation	1.2
Cast impeller, with dye penetrant inspection and x-ray inspection, pore size < 0.060" [1]		1.6
Cast impeller, with dye penetrant inspection and no x-ray inspection		2.8
Composite impeller		3.2

Table 4. Required factor of safety for impact.

Impeller type	Load	n_{impact}
Impeller machined from billet	Impact	1.5
Cast impeller, with dye penetrant inspection and x-ray inspection, pore size < 0.060"		2.0
Cast impeller, with dye penetrant inspection and no x-ray inspection		3.6
Composite		3.9

Table 5. Alternate factor of safety that shall be used in lieu of n_{impact} when impact loads not well defined.

Impeller type	Load	$n_{alternate}$
Impeller machined from billet	Rotation	4.7
Cast impeller, with dye penetrant inspection and x-ray inspection, pore size < 0.060"		6.4
Cast impeller, with dye penetrant inspection and no x-ray inspection		11
Composite impeller		12
Impeller machined from billet with pressed hub insert	Residual stress @ hub insert	4.6
Cast impeller, with cast-in hub insert, with x-ray inspection, pore size < 0.060"		6.5
Cast impeller, with cast-in hub insert, without x-ray inspection		12
Composite impeller		13

6. CONCLUSIONS

Intensive engineering development and initial production work on the Annapolis-design fans lasted from February, 2000 until the first delivery in October, 2000. Most development work focused on design and manufacture of the relatively complicated fan rotor and stator assembly. It was found that the stator design could be simplified while still maintaining the required performance. The rotor remains the

most unique characteristic of the Annapolis design, and as such holds the greatest potential to increase cost. Performance measurements and cost analyses performed as a part of this development show the new Annapolis-design CPS fans have the following costs and benefits:

- Minimal cost premium over previous CPS fan designs.[3]
- 13 dB (re 10^{-12} watt) sound power level reduction compared to previous designs. This reduction leads to a 75% to 88% reduction in perceived noise level compared to previous fans.
- 20% to 23% efficiency improvement compared to previous fans.
- Acceptable flow and stall performance.
- Robust aerodynamic design which can be manufactured using a variety of methods.
- Robust mechanical design that meets current FOS requirements.

In addition to the production of the new fans, new factor of safety recommendations have been developed. A proposed amendment to the performance specification governing the CPS fans has been written and submitted for review.

ACKNOWLEDGMENTS

The authors would like to acknowledge Mr. Michael Slipper and Dr. Yu-Tai Lee of NSWC CD, who originally developed the aerodynamic designs of these fans, for their support in the development of the production fans. Our thanks also go to Mr. Bill Gruber and the employees of American Fan and Woods Air Movement for their hard work in putting the fans into production. Finally, we thank Mr. Dale Sisson of NSWC DD who not only supported the project, but installed the production fans on US Navy ships.

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